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RESEARCH MEMORANDUM

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II - CORRELATION OF HEAT-TRANSFER AND FRICTION DATA

FOR AIR FLOWING IN INCONEL TUBE WITH ROUNDED ENTRANCE

By Warren H. Lowdermilk and Milton D. Grele

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NATIONAL ADVISORY COMMITTEE
FOR AERONAUTICS

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SUMMARY

A heat-transfer investigation, which was an extension of a previously reported NACA investigation, was conducted with air flowing through an electrically heated Inconel tube with a rounded entrance, an inside diameter of 0.402 inch, and a length of 24 inches over a range of conditions, which included Reynolds numbers up to 500,000, average surface temperatures up to 2050° R, and heat-flux densities up to 150,000 Btu per hour per square foot.

Conventional methods of correlating heat-transfer data wherein properties of the air were evaluated at the average bulk, film, and surface temperatures resulted in reductions of Nusselt number of about 38, 46, and 53 percent, respectively, for an increase in surface temperature from 605° to 2050° R at constant Reynolds number. A modified correlation method in which the properties of air were based on the surface temperature and the Reynolds number was modified by substituting the product of the density at the inside tube wall and the bulk velocity for the conventional mass flow per unit cross-sectional area, resulted in a satisfactory correlation of the data for the extended ranges of conditions investigated.

When the friction factors accompanying heat transfer were plotted against conventional Reynolds number, a separation of the data with surface temperature was obtained; an increase in surface temperature resulted in a decrease in friction factor. A somewhat improved correlation of the friction data was obtained by plotting a modified friction factor, which was calculated from a dynamic pressure based on density at the surface against Reynolds number modified in the same manner as in the heat-transfer correlation.

A comparison of the friction and heat-transfer data indicated fair agreement at high Reynolds numbers between the modified half-friction factor and the product of the modified Stanton number and Prandtl number raised to the 0.6 power.

INTRODUCTION

An experimental investigation has been undertaken at the NACA Lewis laboratory to obtain surface-to-fluid heat transfer and associated pressure-drop information over a wide range of surface temperatures and heat-flux densities.

As part of this general program, an investigation is being conducted with air flowing through an electrically heated Inconel tube having an inside diameter of 0.402 inch and an effective heat-transfer length of 24 inches. The results of a preliminary investigation conducted with a rounded entrance to the tube over ranges of Reynolds number from 10,000 to 250,000, average tube-wall temperatures from 680° to 1700° R, heat-flux densities up to 100,000 Btu per hour per square foot, and tube-exit Mach numbers up to 1.0 are reported in reference 1.

The difference between the average tube-wall temperature and the average air temperature had a pronounced effect on the heat-transfer coefficient, as shown in reference 1; consequently, the data could not be correlated when plotted in the conventional manner (that is, Nusselt number divided by Prandtl number to a given power plotted against Reynolds number). By a modification of the conventional Reynolds number and by an evaluation of the properties of the fluid at the temperature of the tube wall, however, the data could be made to fall on a single curve.

The results of an investigation over a greater range of Reynolds numbers and a more extensive discussion of the correlation of the friction data than given in reference 1 are presented herein. The conditions investigated are Reynolds numbers from 7000 to 500,000, average tube-wall temperatures from 605° to 2050° R, and heat-flux densities up to 150,000 Btu per hour per square foot.

In addition to heat-transfer data, the associated static-pressure drops were obtained, from which friction factors have been computed in the conventional manner and plotted against Reynolds number. A modified friction factor has also been computed and is plotted against the modified Reynolds number in a manner consistent with the heat-transfer data.

APPARATUS

The experimental setup used in this investigation is described in detail in reference 1. For convenience, however, the apparatus is briefly reviewed.

Arrangement of Apparatus

A schematic diagram of the heater tube and associated equipment
is shown in figure 1. Compressed air is supplied through a pressure-
regulating valve, a cleaner, and a preheater to a large surge tank.
From the surge tank, the air passes through a second pressure-
regulating valve and a rotameter into a calming tank and then through
the heater tube into a mixing tank from which it is discharged to
the atmosphere.

Electric power is supplied to the heater tube from a 208-volt,
60-cycle supply line through an autotransformer and a 20:1 power
transformer. The low voltage leads of the power transformer are
connected to the heater-tube flanges by copper cables. The capacity
of the electric equipment is 15 kilovolt-amperes.

Heater Tube

After failure of a number of tube-wall thermocouples and pressure
taps had occurred with the original heater tube used in reference 1,
a duplicate test section was fabricated. A diagram of the heater
tube is shown in figure 2. The test section consists of an Inconel
tube having an inside diameter of 0.402 inch, a wall thickness of
0.049 inch, and a total length of 24.75 inches (reference 1). Steel
flanges welded to the tube near each end provide a means of electric
connection to the power source. The distance (24 in.) between the
outer faces of the flanges is considered to be the effective heat-
transfer length. An A.S.M.E. long-radius flow nozzle at the entrance
to the heater tube serves as a smooth entrance and is also used to
measure the air flow. The heater tube is thermally insulated, as
shown in figure 2. The instrumentation of the heater tube is the
same as in reference 1.

PROCEDURE

The experimental procedure was as follows: The air flow through
the heater tube was controlled by varying the pressure in the calming
tank. The entrance-air pressure was adjusted to the value giving
the minimum desired air flow (minimum Reynolds number) with the power
input adjusted to give the approximate desired tube-wall temperature.
After equilibrium conditions had been maintained for approximately
1/2 hour, all power-input, pressure, and temperature readings were
recorded. The entrance-air pressure and power input were then
increased to increase the Reynolds number and to maintain the desired
tube-wall temperature and the foregoing procedure was repeated.

Data were obtained over a range of Reynolds numbers from 7000 to 500,000, average inside-tube-wall temperatures from 605° to 2050° R (corresponding maximum local wall temperatures 610° to 2210° R) and heat-flux densities up to 150,000 Btu per hour per square foot of heat-transfer area. The air temperature at the tube entrance was about 545° R and the air pressure varied from 15 to 90 pounds per square inch absolute.

SYMBOLS

The following symbols are used in this report:

A cross-sectional area, (sq ft)

c_p specific heat of air at constant pressure, $(Btu/(lb)(°F))$

D inside diameter of heater tube, (ft)

f average friction factor

G mass flow per unit cross-sectional area, $(lb/(hr)(sq\ ft))$

g acceleration due to gravity, 4.17×10^8 (ft/hr^2)

h average heat-transfer coefficient, $(Btu/(hr)(sq\ ft)(°F))$

k thermal conductivity of air, $(Btu/(hr)(sq\ ft)(°F/ft))$

k_I thermal conductivity of Inconel, $(Btu/(hr)(sq\ ft)(°F/ft))$

L effective heat-transfer length of heater tube, (ft)

P total pressure, (lb/sq ft absolute)

p static pressure, (lb/sq ft absolute)

Δp over-all static-pressure drop across heater tube, (lb/sq ft)

Δp_f friction static-pressure drop across heater tube, (lb/sq ft)

Q rate of heat transfer to air, (Btu/hr)

R gas constant for air, 53.35 $(ft-lb/(lb)(°F))$

S   heat-transfer area of heater tube, 0.211 (sq ft)

T   total temperature, ($^\circ$R)

t   static temperature, ($^\circ$R)

V   velocity, (ft/hr)

W   air flow, (lb/hr)

γ ratio of specific heats of air

μ absolute viscosity of air, (lb/(hr)(ft))

ρ density of air, (lb/cu ft)

$c_p\mu/k$ Prandtl number

DG/μ Reynolds number

$\rho_s V_b D/\mu_s$ modified Reynolds number

hD/k Nusselt number

$h/c_p G$ Stanton number

$h/c_{p,s}\rho_s V_b$ modified Stanton number

Subscripts:

1   heater-tube entrance

2   heater-tube exit

o   outer surface of heater tube

b   physical properties of air evaluated at average bulk
                (total) temperature T_b

f   physical properties of air evaluated at average film
                temperature T_f

s   physical properties of air evaluated at average inside-
                wall temperature T_s

METHOD OF CALCULATION

The method of calculating the experimental data was essentially the same as described in reference 1. The important details of the method and pertinent equations will, however, be repeated for convenience.

Heat-transfer coefficients. - The average heat-transfer coefficient h was computed from the experimental data by the relation

$$h = \frac{Wc_{p,b}(T_2 - T_1)}{S(T_s - T_b)} \tag{1}$$

In equation (1), the bulk temperature of the air T_b was taken as the arithmetic mean of the total temperatures T_1 and T_2 measured in the inlet and outlet tanks, respectively.

The temperature of the inside tube surface T_s was obtained by plotting curves of outside-tube-wall temperature against distance along the tube, measuring the area under the curve, dividing by the tube length, and subtracting the computed temperature drop through the tube wall. The equation used to relate the outside- and inside-tube-wall temperatures is

$$T_s = T_o - 0.0092 \frac{Q}{k_I} \tag{2}$$

Equation (2) was obtained from references 1 and 2.

Because of lack of information on the variation of the thermal conductivity of Inconel with temperature, a constant value was assumed (reference 1) in computing the temperature drop through the tube wall. Limited data on the variation of the thermal conductivity of Inconel with temperature became available and were accordingly used herein. The use of variable thermal conductivity resulted in a maximum change in the temperature difference between the tube wall and the air $(T_s - T_b)$ of about 1 percent from that obtained with constant conductivity.

Correlation of heat-transfer data. - The heat-transfer data are first correlated in the conventional manner by means of the Nusselt equation, which is

$$\frac{hD}{k} = C\left(\frac{DG}{\mu}\right)^m \left(\frac{c_p\mu}{k}\right)^n \tag{3}$$

where

C,m   constants determined from experimental results

n   exponent of Prandtl group $\dfrac{c_p\mu}{k}$, taken as 0.4

The physical properties of the air are evaluated at (1) average
bulk temperature T_b, (2) average film temperature T_f, and
(3) average inside-tube-wall temperature T_s.

The average film temperature T_f was taken as the arithmetic
mean of the average bulk total temperature T_b and the average
inside-tube-wall temperature T_s. The physical properties of air
were obtained from reference 3.

When the data are correlated by any of the methods previously
described, a definite effect of the temperature difference between
the tube wall and the air is evident, as shown in reference 1.
This effect of temperature difference was eliminated by evaluating
the air properties at the surface temperature of the tube and
modifying the Reynolds number by substituting the product of air
density evaluated at the surface temperature ρ_s and the average
air velocity evaluated at the bulk temperature V_b for the con-
ventional mass flow per unit cross-sectional area G or $\rho_b V_b$.
Hence, the heat-transfer data of the present investigation are
correlated secondly by the relation

$$\left(\frac{hD}{k_s}\right) = C_1 \left(\frac{\rho_s V_b D}{\mu_s}\right)^{m_1} \left(\frac{c_{p,s}\,\mu_s}{k_s}\right)^{0.4} \tag{4}$$

The relation between the modified and the conventional Reynolds
number is given by the following equation:

$$\frac{\rho_s V_b D}{\mu_s} = \left(\frac{DG}{\mu_s}\right)\left(\frac{\rho_s}{\rho_b}\right) = \left(\frac{DG}{\mu_s}\right)\left(\frac{T_b}{T_s}\right) = \left(\frac{DG}{\mu_b}\right)\left(\frac{T_b}{T_s}\right)\left(\frac{\mu_b}{\mu_s}\right) \tag{5}$$

The value of ρ_b was computed by use of T_b, which is a total rather than a static temperature.

Friction factors. - The friction static-pressure drop Δp_f was obtained by subtracting the momentum-pressure drop from the measured drop in static pressure across the tube. Thus

$$\Delta p_f = \Delta p - \frac{W}{gA}(V_2 - V_1) = \Delta p - \left(\frac{W}{A}\right)^2 \frac{R}{g}\left(\frac{t_2}{p_2} - \frac{t_1}{p_1}\right) \tag{6}$$

where

$$t_1 = T_1 \left(\frac{p_1}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \tag{7}$$

and

$$t_2 = -\frac{\gamma g}{(\gamma-1)R}\left(\frac{p_2 A}{W}\right)^2 + \sqrt{\left[\frac{\gamma g}{(\gamma-1)R}\left(\frac{p_2 A}{W}\right)^2\right]^2 + 2T_2\left[\frac{\gamma g}{(\gamma-1)R}\left(\frac{p_2 A}{W}\right)^2\right]} \tag{8}$$

In equation (7), the total pressure at the tube entrance P_1 was assumed to be equal to the static pressure in the inlet tank, where the air velocity was negligibly small.

An average friction factor f_b was calculated from the friction-pressure drop by use of the conventional relation

$$f_b = \frac{\Delta p_f}{4\frac{L}{D}\frac{\rho_b V_b^2}{2g}} = \frac{2g\rho_b\Delta p_f}{4\frac{L}{D}\left(\frac{W}{A}\right)^2} \tag{9}$$

where the average bulk density ρ_b was taken as

$$\rho_b = \frac{1}{R}\left(\frac{p_1 + p_2}{t_1 + t_2}\right) \tag{10}$$

A modified friction factor f_s was also calculated from the relation

$$f_s = \frac{\Delta p_f}{4\frac{L}{D}\frac{\rho_s V_b^2}{2g}} = f_b\left(\frac{\rho_b}{\rho_s}\right) = f_b\left(\frac{T_s}{t_b}\right) \tag{11}$$

where the dynamic pressure is based on air density evaluated at the average surface temperature and the average bulk static temperature t_b was taken as the arithmetic mean of the entrance and exit static temperatures as computed by means of equations (7) and (8), respectively.

In equations (9) and (11), the assumption is made that the variation in average velocity V along the length of the heater tube is small. For a large variation in average velocity, the friction coefficient is more accurately given by

$$f = \frac{\Delta p_f}{\frac{4}{2g} D \int_0^L \rho V^2 \, dL} \tag{11a}$$

The integral corresponds to an effective mean value of velocity greater than V_b. Inasmuch as the variation in average velocity is unknown in the present investigation, the preceding relation cannot be integrated.

The friction data are first correlated in the conventional manner by plotting one-half the friction factor $f_b/2$ against conventional Reynolds number DG/μ_b, where f_b was calculated from equation (9). The data were then correlated in a modified manner consistent with the heat-transfer data by plotting one-half the modified friction factor $f_s/2$ against modified Reynolds number $\rho_s V_b D/\mu_s$, where f_s was calculated from equation (11).

RESULTS AND DISCUSSION

Heat Balance

The heat balance obtained in this investigation is shown in figure 3 where the electric heat input minus the external heat loss is plotted against the rate of heat transfer to the air as determined from the air flow, the specific heat, and the temperature

rise. The external heat losses for the new heater tube were found
by check runs to be the same as shown in reference 1. A good heat
balance is indicated with a maximum deviation of approximately
5 percent from the match line.

Correlation of Heat-Transfer Coefficients

Correlation based on bulk temperature. - The correlation of
the heat-transfer coefficients in which the physical properties of
the air are evaluated at the average bulk temperature is shown in
figure 4(a), where the Nusselt number divided by Prandtl number
raised to the 0.4 power $(hD/k_b)/(c_{p,b}\mu_b/k_b)^{0.4}$ is plotted against
Reynolds number DG/μ_b. Included for comparison is the average
line (dash-dot-dash) obtained by McAdams from correlation of the
results of various investigators (reference 4, p. 168) and the
average line (dashed) obtained by Drexel and McAdams (reference 5)
for the data of the same investigators but using the same physical
properties used herein. The line obtained by Drexel and McAdams
is therefore considered preferable for comparison with the correla-
tion of this investigation. In order to avoid congestion, none of
the data from reference 1 are included in this plot, however, all
these data are in good agreement with reference 1, as will be seen
in subsequent figures.

A family of parallel lines for the different temperature levels
is obtained with slopes of 0.8 at Reynolds numbers above about
25,000. Similar results are shown in reference 1. At lower
Reynolds numbers, the data decrease indicating the presence of
transitional-flow conditions resulting from the rounded entrance.
The low-temperature data (surface temperature, 605° to 680° R) are
in good agreement with the line of reference 5; the high-temperature-
data lines progressively decrease with increased wall temperature
and corresponding increased temperature difference between wall and
air.

Correlation based on film temperature. - The data are replotted
in figure 4(b) with the physical properties of the air evaluated at
the average film temperature. A separation of the data with temper-
ature level similar to that shown in figure 4(a) is obtained. The
separation is actually slightly greater as will be seen in a subse-
quent figure.

Correlation based on surface temperature. - The data are plotted
in figure 4(c) with the physical properties of the air evaluated at
the average surface temperature. Again the data show a separation
with temperature level.

In order to show better the magnitude of the temperature effect
on the heat-transfer correlations, the data from figure 4 are cross-
plotted in figure 5 against the temperature difference between the
tube wall and the air for a Reynolds number of 100,000. An increase
in the temperature difference from 55° to 1230° F results in a
decrease in Nusselt number divided by Prandtl number raised to the
0.4 power of 38, 46, and 53 percent when the physical properties are
evaluated at the bulk, film, and surface temperatures, respectively.

Correlation based on modified Reynolds number and surface
temperature. - The Reynolds number modified by substituting the
product of density evaluated at the surface temperature ρ_s and
velocity evaluated at the bulk temperature V_b for the conventional
mass flow per unit cross-sectional area G and all physical prop-
erties evaluated at the inside-tube-wall or surface temperature are
shown in figure 6. The data obtained in reference 1 with the orig-
inal heater tube are included in the figure and are in good agree-
ment with the data obtained with the duplicate heater tube used in
this investigation.

A good correlation of all the data for a range of average wall
temperatures from 605° to 2050° R and the corresponding temperature
difference between wall and air from 55° to 1230° F is obtained.
The equation of the line, which best represents all the data for
modified Reynolds numbers above approximately 10,000, is

$$\left(\frac{hD}{k_s}\right)\bigg/\left(\frac{c_{p,s}\mu_s}{k_s}\right)^{0.4} = 0.023\left(\frac{\rho_s V_b D}{\mu_s}\right)^{0.8} \tag{12}$$

The constant 0.023 is approximately 2 percent higher than the value
0.022 given in reference 1.

An alternate expression of the preceding equation involving a
modified Stanton number is obtained by dividing both sides of the
equation by the modified Reynolds number, which results in

$$\left(\frac{h}{c_{p,s}\rho_s V_b}\right)\left(\frac{c_{p,s}\mu_s}{k_s}\right)^{0.6} = 0.023\left(\frac{\rho_s V_b D}{\mu_s}\right)^{-0.2} \tag{12a}$$

All the data are replotted neglecting Prandtl number, that is, Nusselt number hD/k_s is plotted against modified Reynolds number $\rho_s V_b D/\mu_s$ in figure 7. Omission of the Prandtl number is seen to have a negligible effect on the correlation. The equation of the line best representing all the data is

$$\frac{hD}{k_s} = 0.019 \left(\frac{\rho_s V_b D}{\mu_s}\right)^{0.8} \tag{13}$$

Here again, the constant 0.019 is slightly higher than the value of 0.018 given in reference 1.

Correlation of Friction Factors

Correlation based on bulk temperature. - The average half-friction factor $f_b/2$ is plotted against the conventional Reynolds number DG/μ_b in figure 8. The friction factor was calculated from equation (9) and the physical properties of the air were evaluated at the bulk temperature. Included for comparison is the line representing an equation often used for evaluating friction factor in the turbulent range of Reynolds numbers,

$$\frac{f}{2} = 0.023 \left(\frac{GD}{\mu}\right)^{-0.2} \tag{14}$$

which was obtained from reference 4 (p. 119). The isothermal and low-temperature data (surface temperature, 545°, 605°, and 680° R) are in fair agreement with the reference line; the high-temperature-data lines, however, progressively decrease with increasing surface temperature. Similar trends (not shown) are obtained when the viscosity term in the Reynolds number is evaluated at either the film or surface temperature.

Correlation based on modified Reynolds number and surface temperature. - The modified half-friction factor $f_s/2$ (as obtained from equation (11)) is plotted against the modified Reynolds number in figure 9. The data of reference 1 are included for comparison. The line corresponding to the reference equation in figure 8 is also included.

Although the data again show some scatter, the correlation is better than in figure 8, particularly in the high Reynolds number

region. The scatter in the low Reynolds number region may be a reflection of delayed transition from laminar to turbulent flow resulting from the rounded tube entrance.

Comparison of Friction and Heat-Transfer Coefficients

A comparison between the friction and heat-transfer coefficients is shown in figure 10 where the half-friction factor $f_s/2$ from figure 9 and the modified Stanton number multiplied by Prandtl number to the 0.6 power $\left(\dfrac{h}{c_{p,s}\rho_s V_b}\right)\left(\dfrac{c_{p,s}\mu_s}{k_s}\right)^{0.6}$ are plotted against modified Reynolds number $\rho_s V_b D/\mu_s$. The friction and heat-transfer parameters are in fair agreement, particularly in the high Reynolds number region; hence, the friction parameter can be equated to the heat-transfer parameter

$$\frac{f_s}{2} = \left(\frac{h}{c_{p,s}\rho_s V_b}\right)\left(\frac{c_{p,s}\mu_s}{k_s}\right)^{0.6} \tag{15}$$

which by rearrangement indicates that the ratio of the modified Stanton number to the modified half-friction factor is a function of the Prandtl number

$$\frac{\left(\dfrac{h}{c_{p,s}\rho_s V_b}\right)}{\left(\dfrac{f_s}{2}\right)} = \left(\frac{c_{p,s}\mu_s}{k_s}\right)^{-0.6} \tag{15a}$$

This relation parallels one of the conventional skin-friction - heat-transfer analogy relations wherein the following equations (given in reference 4) for determining the heat-transfer coefficient and friction factor, respectively,

$$\frac{hD}{k} = 0.023 \left(\frac{GD}{\mu}\right)^{0.8} \left(\frac{c_p\mu}{k}\right)^{0.4} \tag{16}$$

$$f = 0.046 \left(\frac{GD}{\mu}\right)^{-0.2} \tag{17}$$

can be combined to give the relation

$$\frac{\dfrac{h}{c_p G}}{\dfrac{f}{2}} = \left(\frac{c_p \mu}{k}\right)^{-0.6}$$ (18)

SUMMARY OF RESULTS

The results of a heat-transfer investigation conducted with air flowing through an electrically heated Inconel tube with a rounded entrance, an inside diameter of 0.402 inch, and a length of 24 inches, over a range of Reynolds numbers from 7000 to 500,000, average inside-tube-wall temperatures from 605° to 2050° R (corresponding maximum local temperatures up to 2210° R), and heat-flux densities up to 150,000 Btu per hour per square foot showed that:

1. Correlation of the average heat-transfer coefficients according to the familiar Nusselt relation, wherein physical properties were evaluated at the average air bulk temperature, resulted in a separation with temperature level of the parameter Nusselt number divided by Prandtl number to the 0.4 power; at constant Reynolds number the parameter decreased about 38 percent with increase in average surface temperature from 605° to 2050° R. Evaluation of the physical properties at the average film and average surface temperatures reduces the parameter by 46 and 53 percent, respectively, for the same increase in temperature.

2. A good correlation of the heat-transfer data was obtained for the entire range of temperatures when the viscosity μ_s, thermal conductivity k_s, and the specific heat $c_{p,s}$ of the air were evaluated at the average surface temperature and the Reynolds number was modified. The equation of the line representing the data for Reynolds numbers above 10,000 is

$$\left(\frac{hD}{k_s}\right) \Big/ \left(\frac{c_{p,s}\mu_s}{k_s}\right)^{0.4} = 0.023 \left(\frac{\rho_s V_b D}{\mu_s}\right)^{0.8}$$

where

D   inside diameter of heater tube

h   average heat-transfer coefficient

V_b average bulk velocity of air

ρ_s density of air evaluated at surface temperature

Satisfactory correlation was also obtained neglecting Prandtl number in which case the equation for the line fitting the data is

$$\frac{hD}{k_s} = 0.019\left(\frac{\rho_s V_b D}{\mu_s}\right)^{0.8}$$

3. Correlation of average friction factor with conventional Reynolds number, wherein the physical properties are evaluated at average bulk air temperature, resulted in a separation of the data with temperature level. The friction factor progressively decreased with increase in surface temperature.

4. Somewhat better correlation of the friction data was obtained when the friction factor was calculated from a dynamic pressure based on air density evaluated at the surface temperature and was plotted against the modified Reynolds number.

5. A comparison of the friction and heat-transfer data indicated fair agreement at high Reynolds numbers between the modified half-friction factor and the product of the modified Stanton number and modified Prandtl number raised to the 0.6 power.

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       National Advisory Committee for Aeronautics,
              Cleveland, Ohio.

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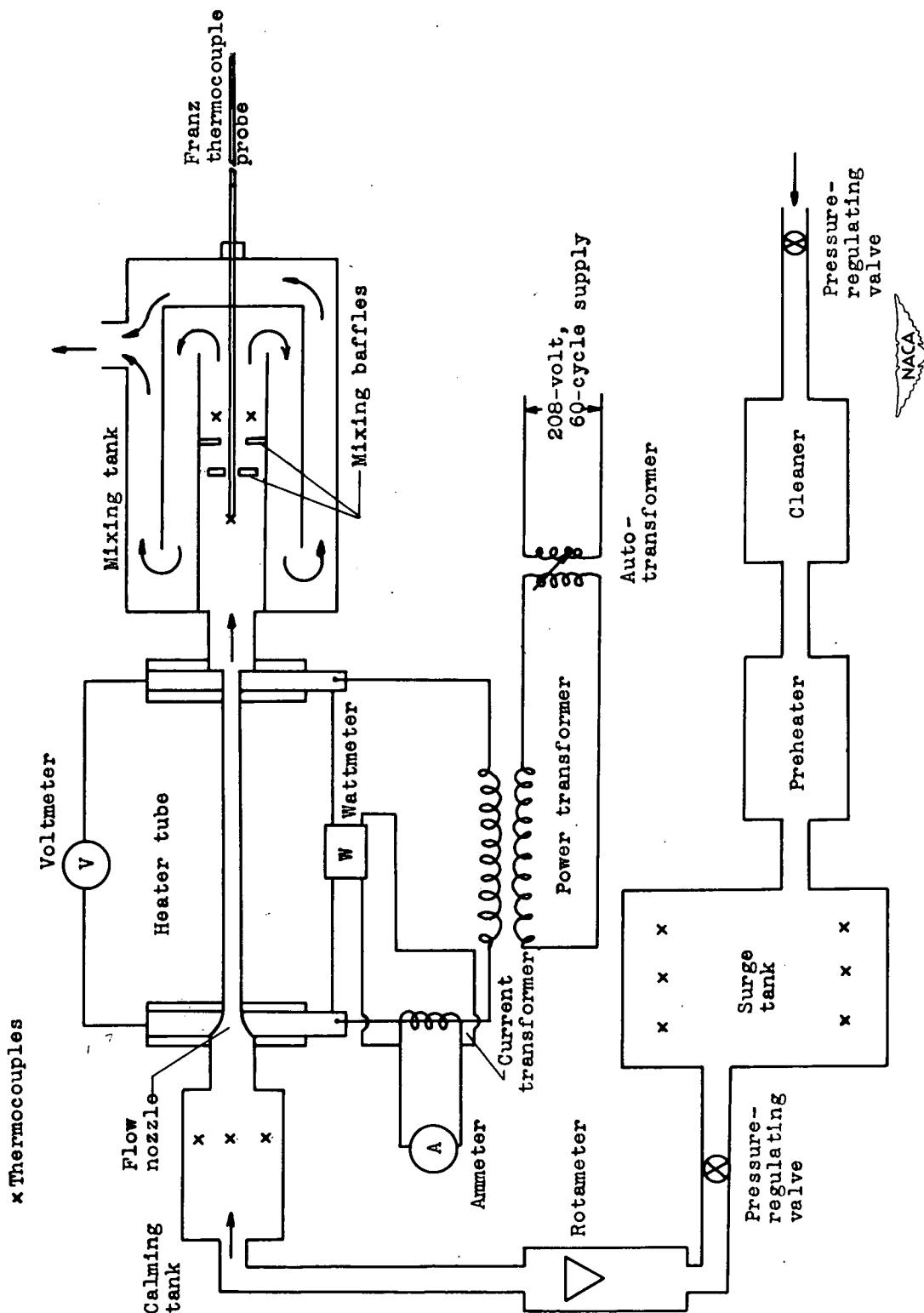
Figure 1. - Schematic diagram showing arrangement of experimental apparatus.
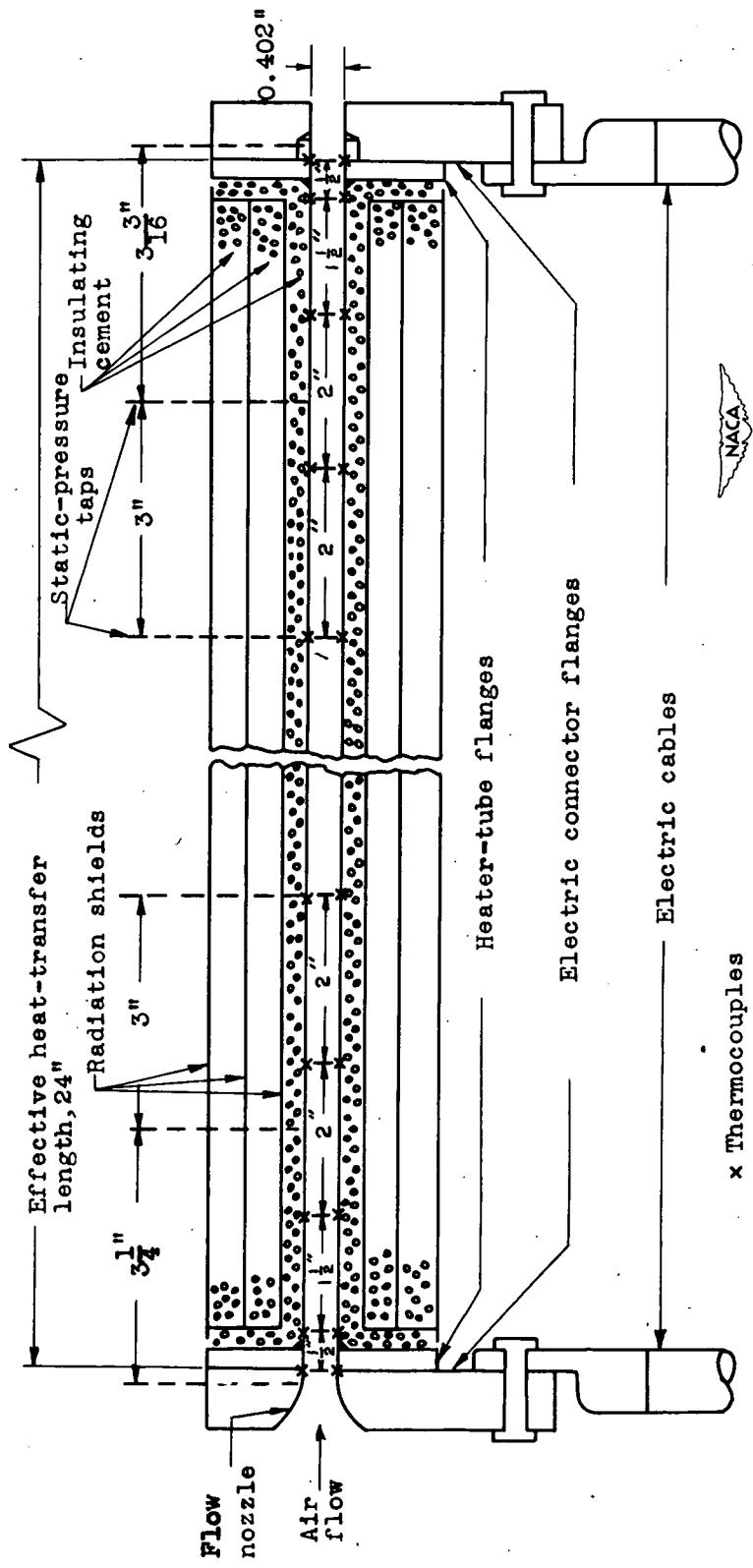
Figure 2. - Schematic diagram of test section showing thermocouple and pressure-tap locations.
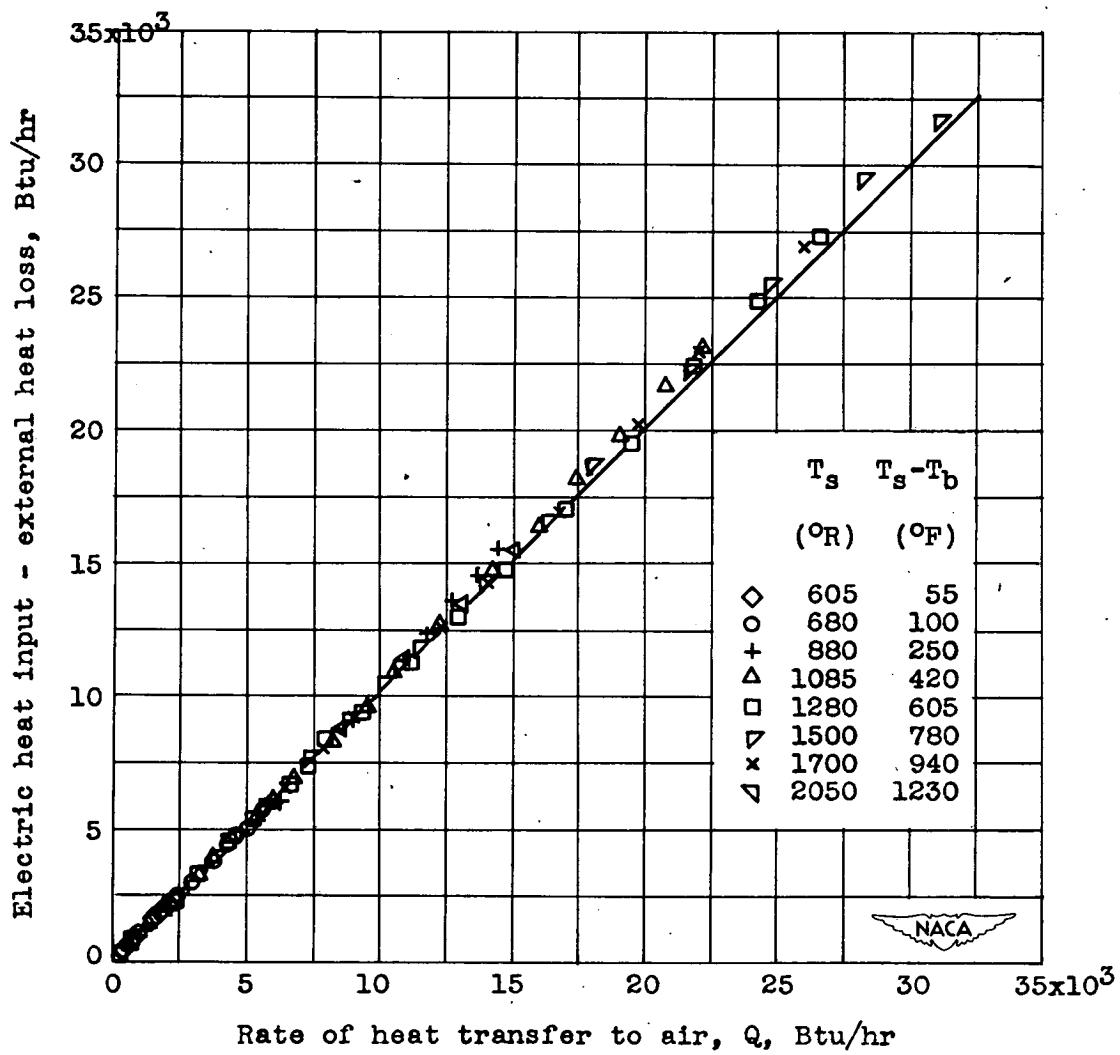
Figure 3. - Heat balance.
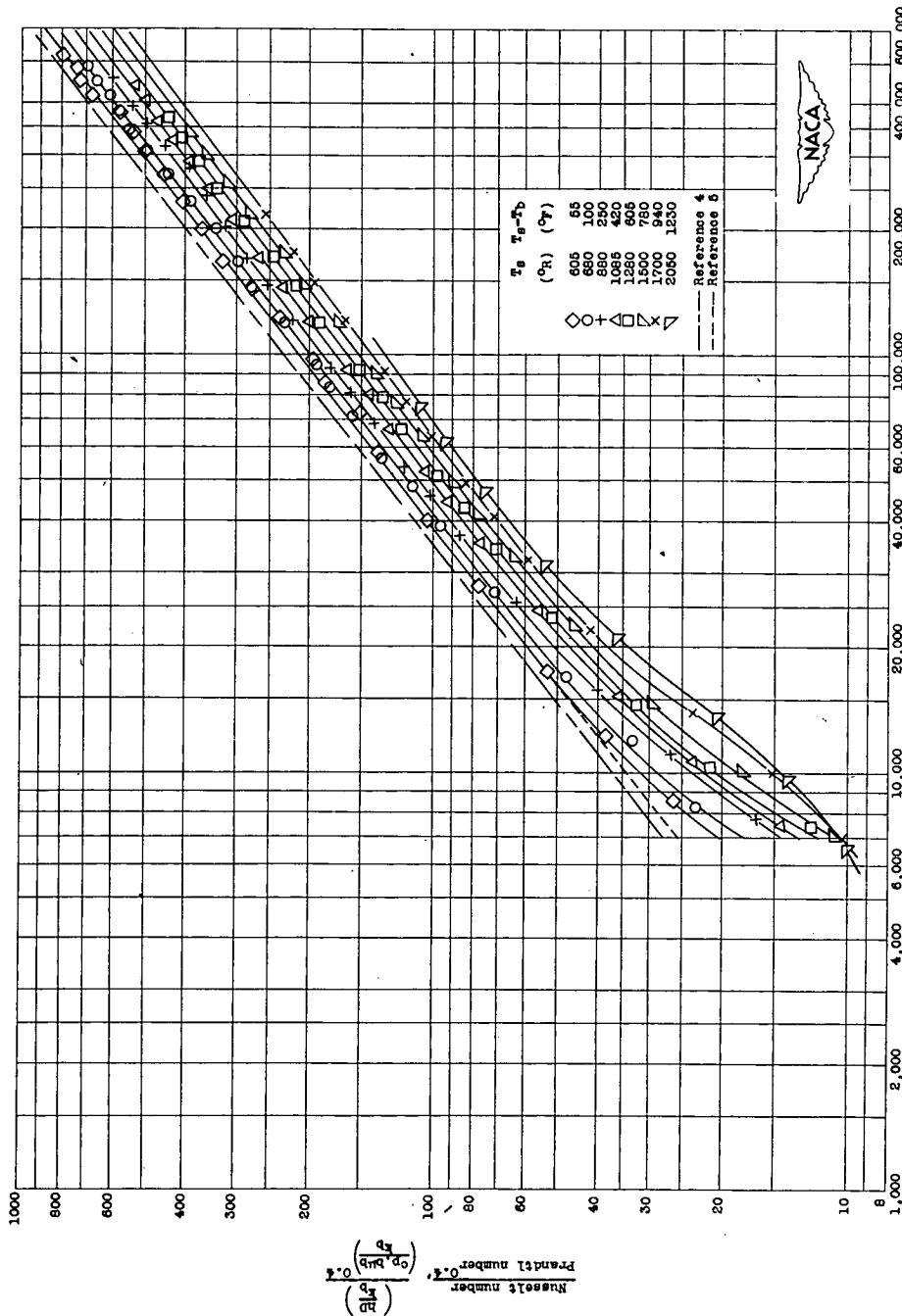
(a) Physical properties of air evaluated at average bulk temperature T_b.

Figure 4. – Correlation of heat-transfer coefficients.
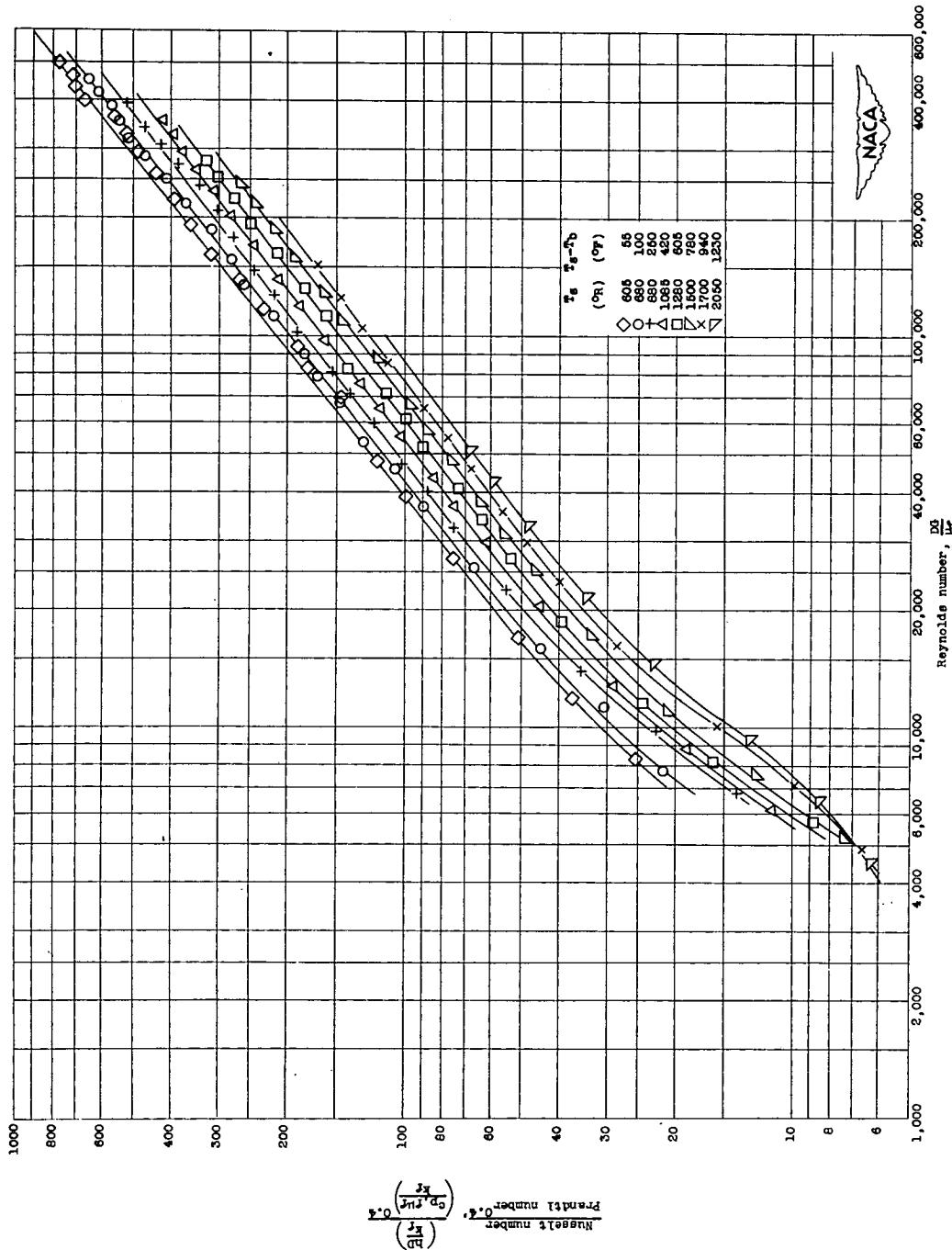
(b) Physical properties of air evaluated at average film temperature T_f.

Figure 4. – Continued. Correlation of heat-transfer coefficients.
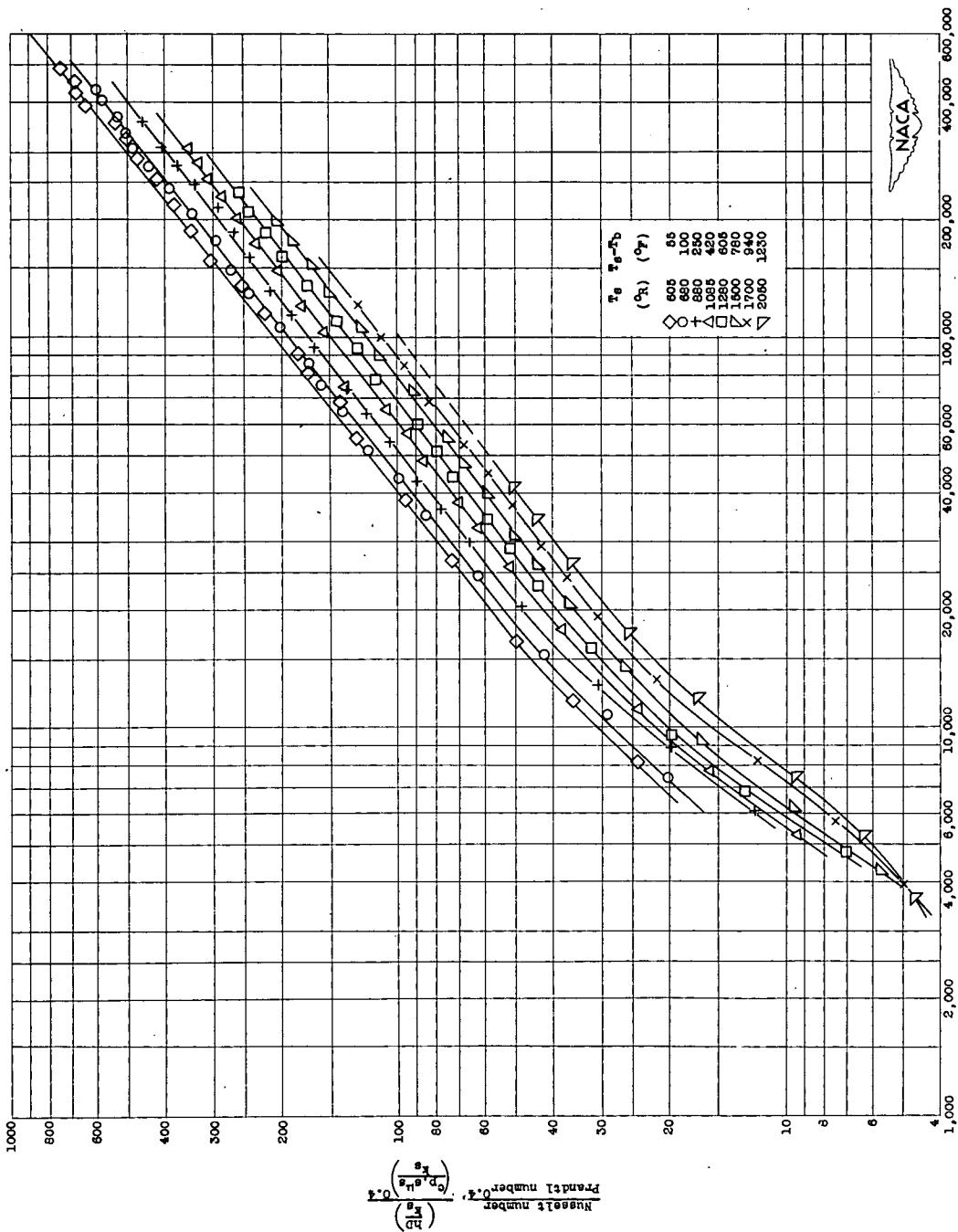
(c) Physical properties of air evaluated at average inside-tube-wall temperature T_s.

Figure 4. – Concluded. Correlation of heat-transfer coefficients.
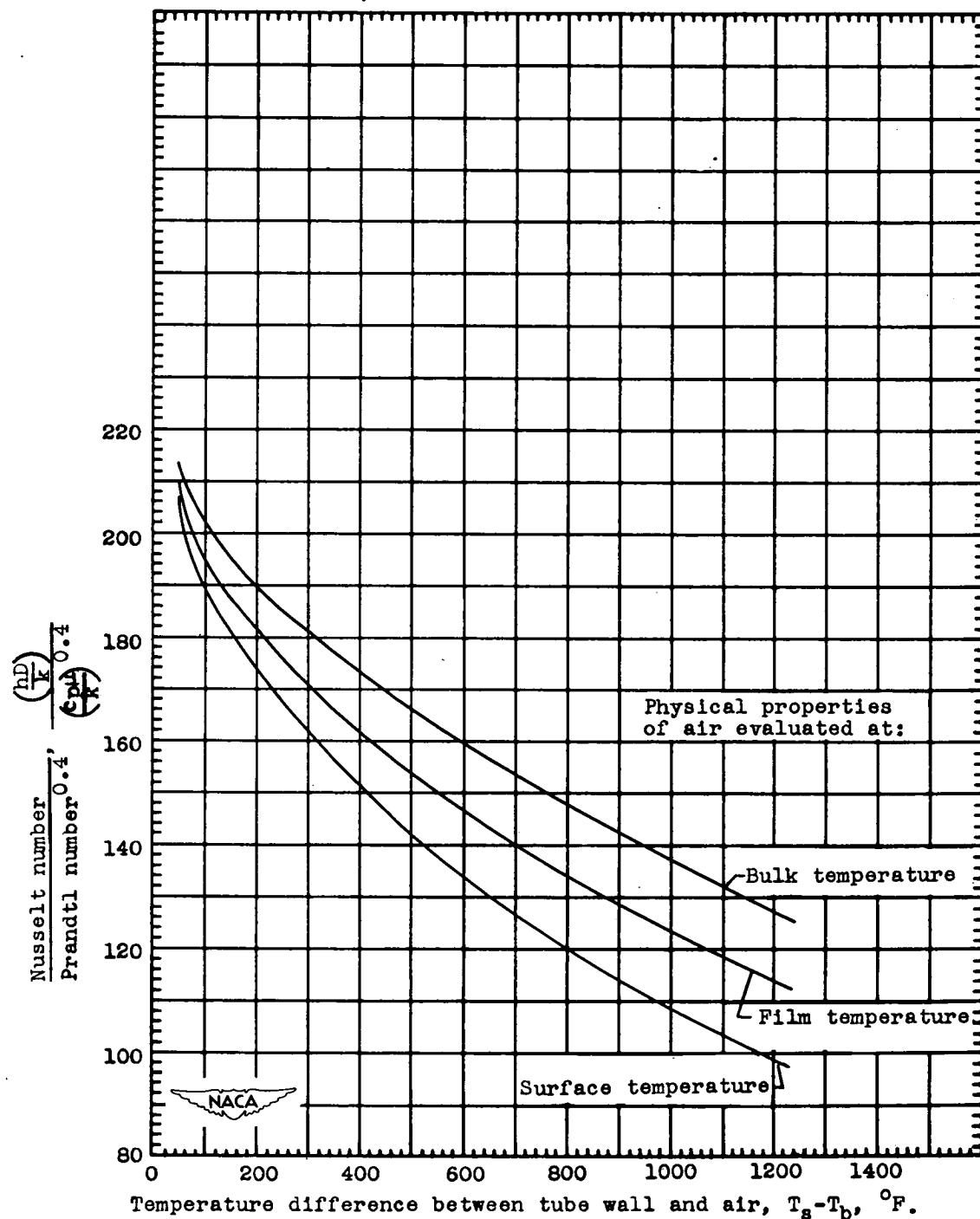
Figure 5. - Variation of Nusselt number divided by Prandtl number
to the 0.4 power with temperature difference between the tube
wall and air for a constant Reynolds number of 100,000.
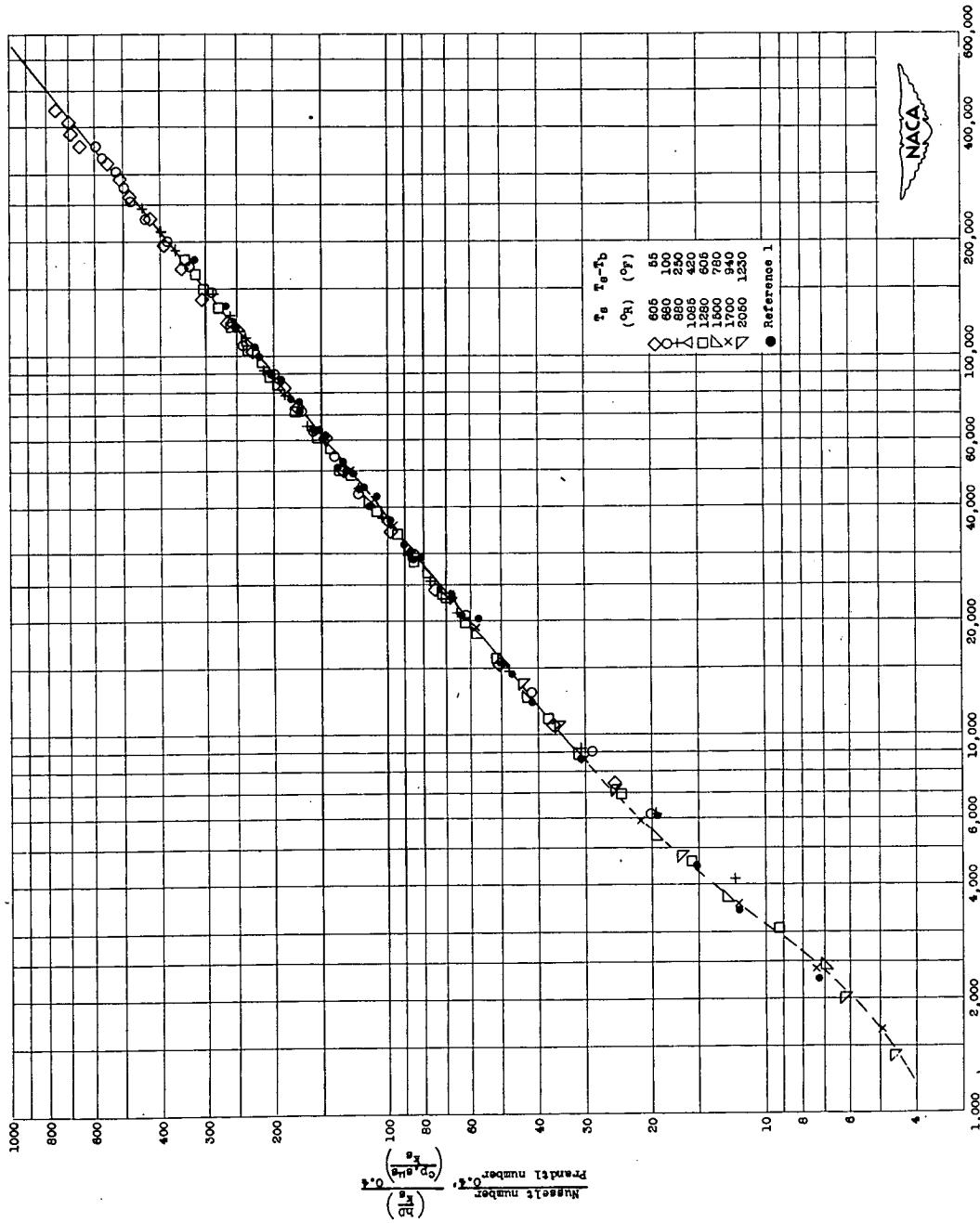
Figure 6. - Correlation of heat-transfer coefficients using a modified Reynolds number. Physical properties of air evaluated at average inside-tube-wall temperature T_s.
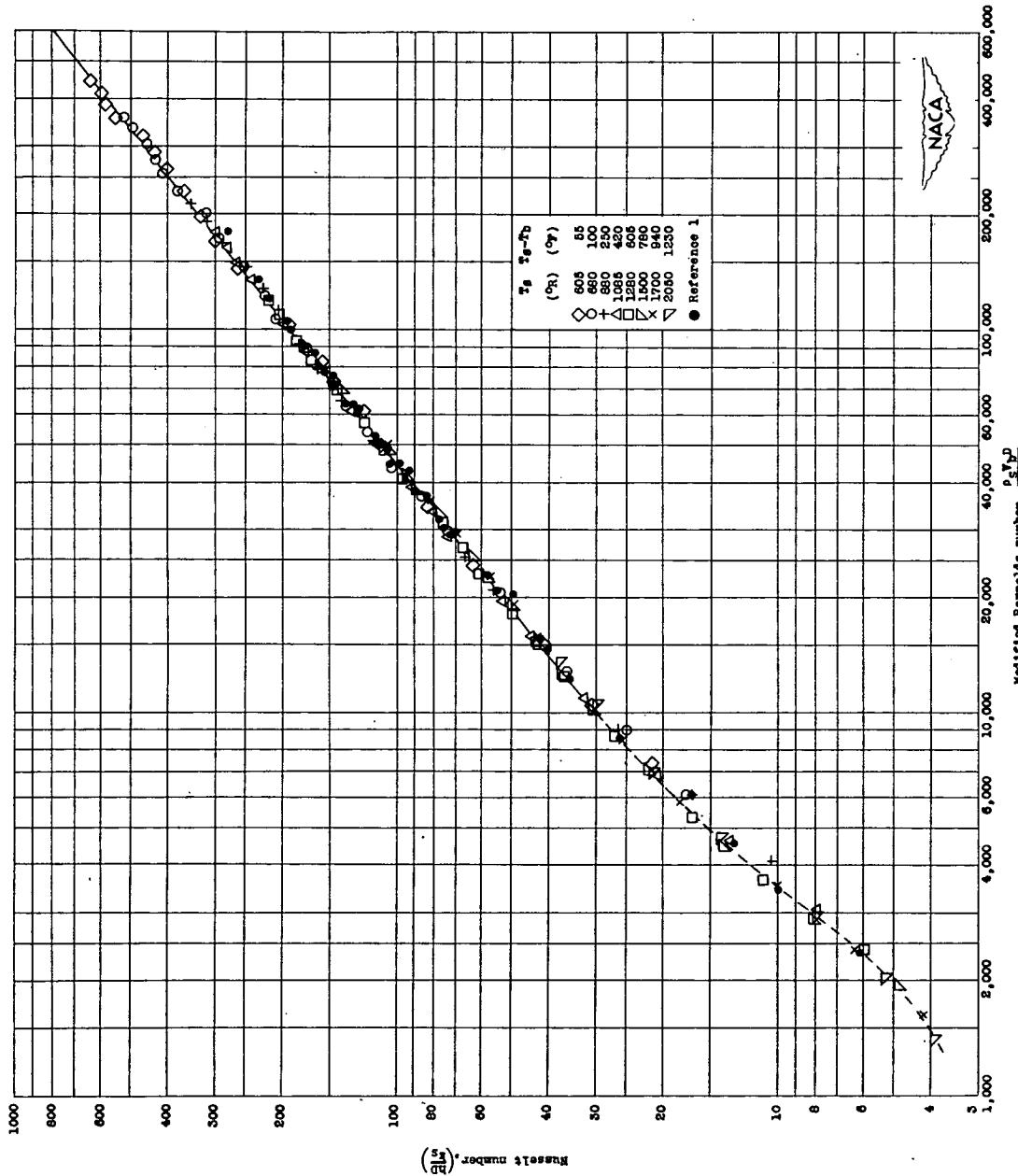
Figure 7. - Correlation of heat-transfer coefficients using a modified Reynolds number and neglecting Prandtl number. Physical properties of air evaluated at average inside-tube-wall temperature T_s.
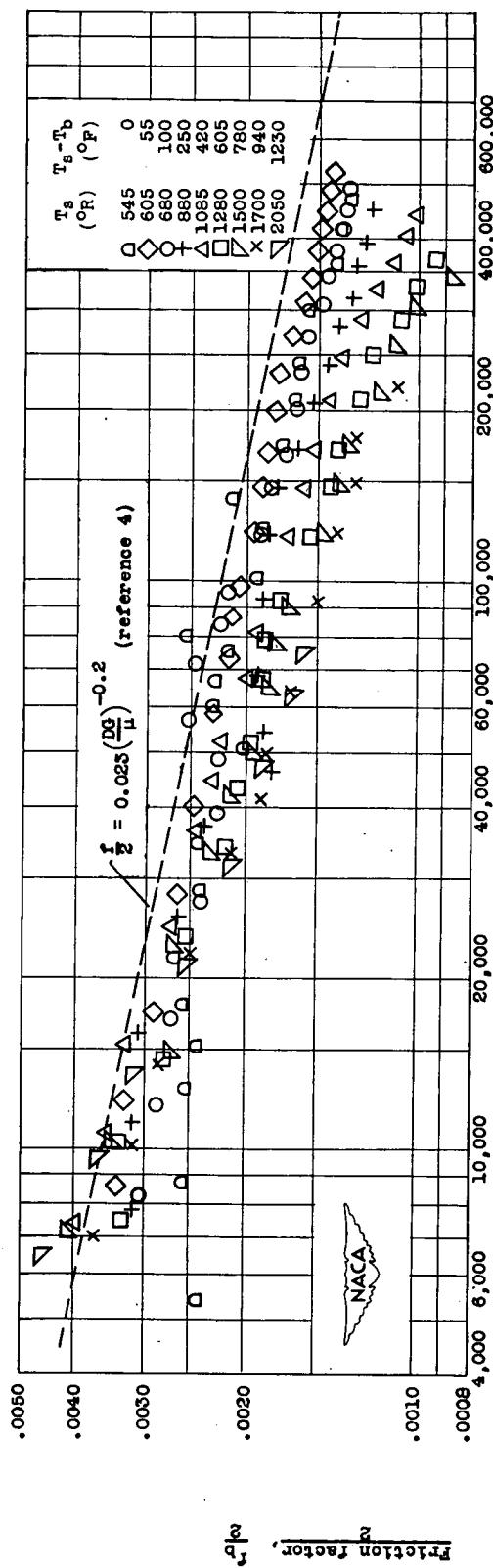
Figure 8. – Correlation of friction factors. Physical properties of air evaluated at average bulk temperature T_b.
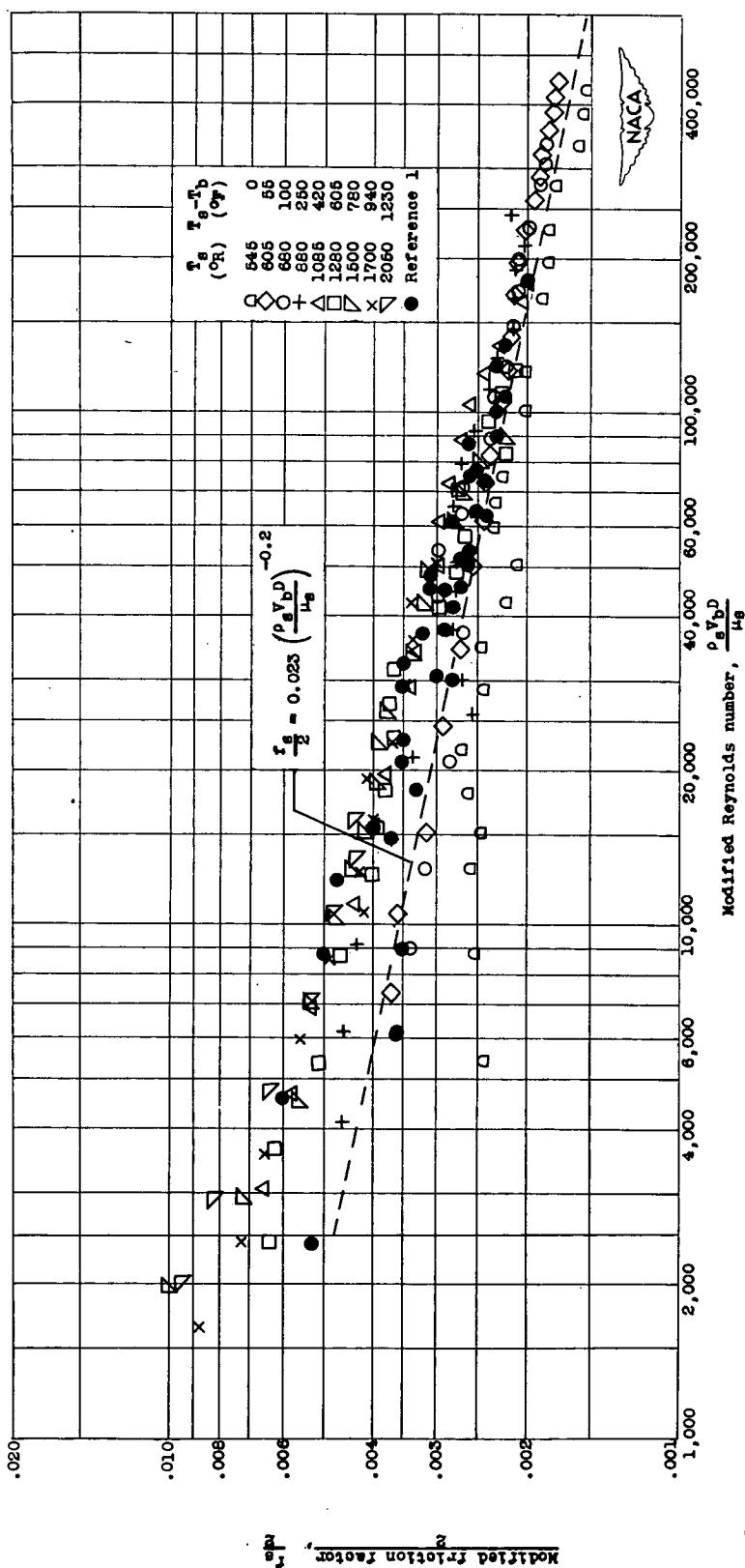
Figure 9. - Correlation of modified friction factors. Physical properties of air evaluated at average inside-tube-wall temperature T_s.
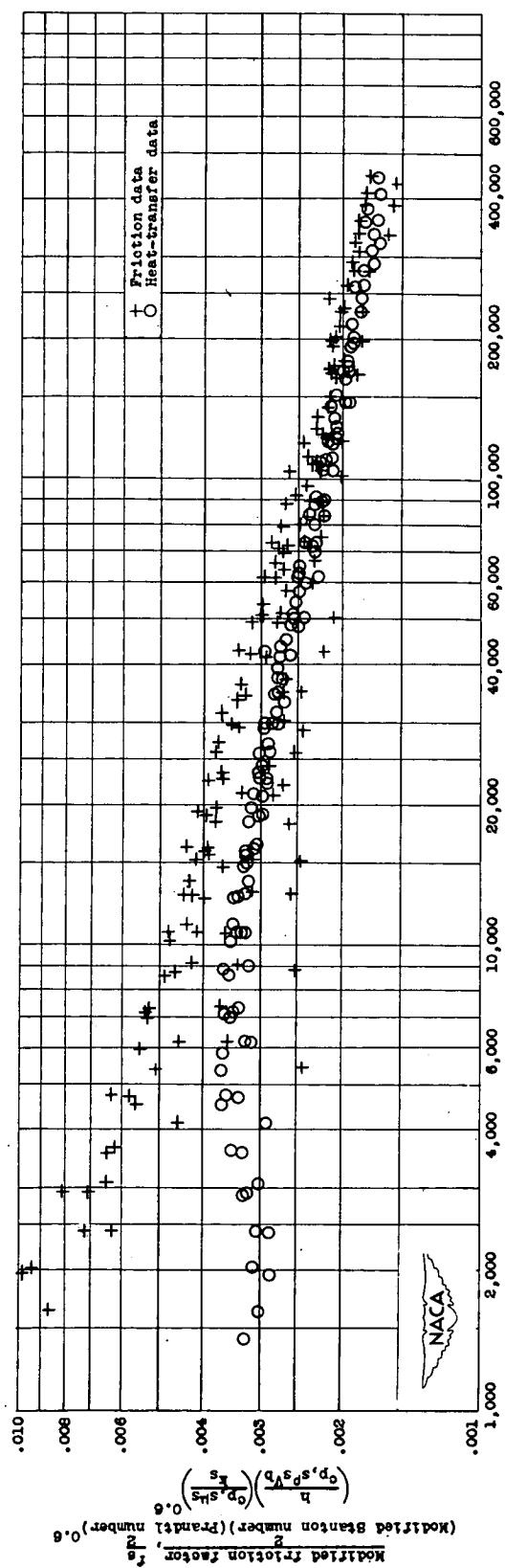
Figure 10. – Comparison of friction factors with heat-transfer coefficients. Physical properties of air evaluated at average inside-tube-wall temperature T_s.